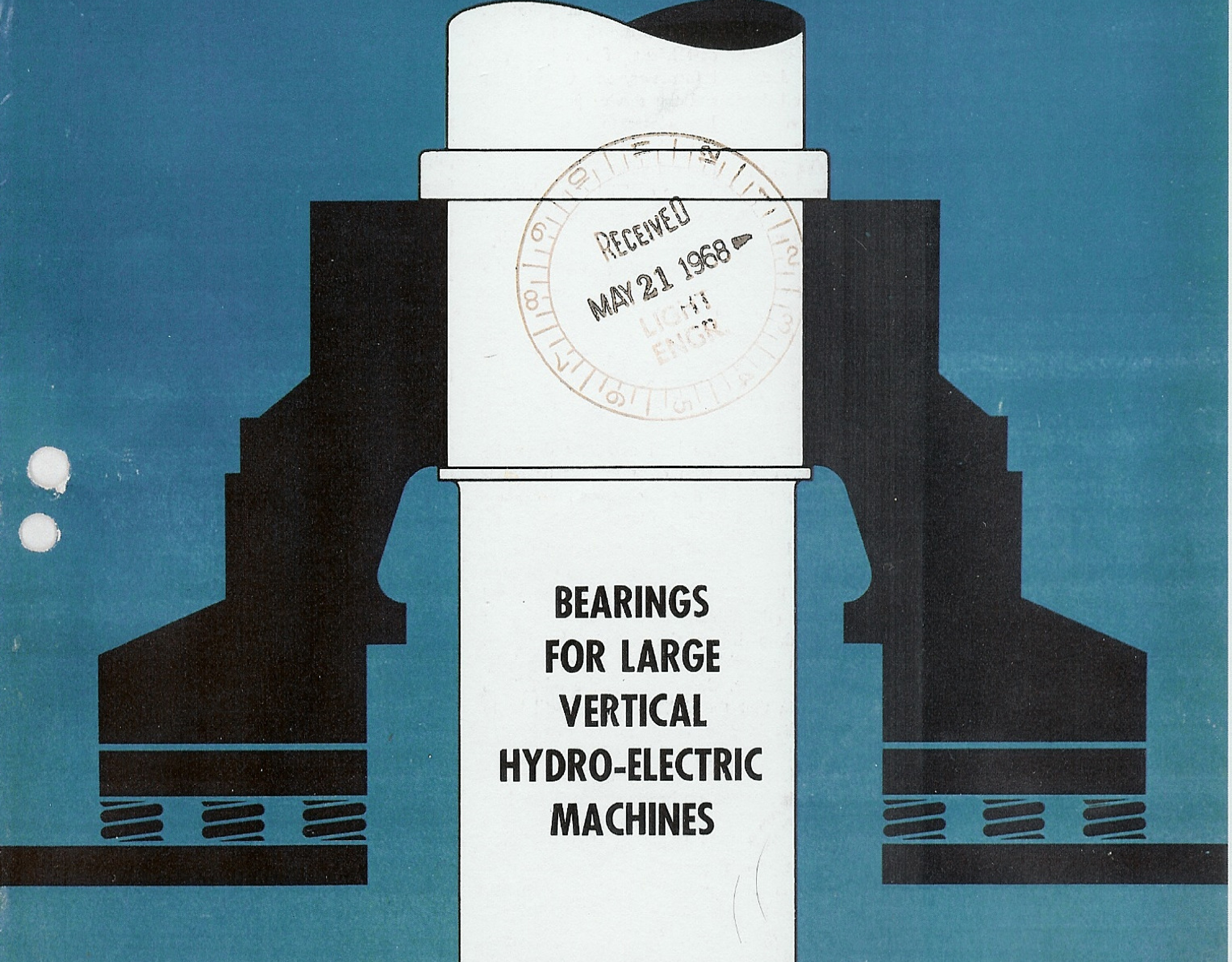
BEARINGS FOR LARGE VERTICAL HYDRO-ELECTRIC MACHINES



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INTRODUCTION

The bearings for large vertical machines are designed as hydrodynamic bearings of the pad type, where the load is transferred from the rotating to the stationary parts via a wegde-shaped oil film. The sliding surfaces are kept completely separate from one another, and the frictional force is dependent only on internal forces in the oil film.

The change of the coefficient of friction and the building up of the oil film are clarified in the diagram in Fig. 1, where the coefficient of friction has been plotted as a function of the dimensionless quantity $\eta\omega/p$.

The mode of operation of the hydrodynamic bearing can be classified in relation to the friction between sliding surfaces in the following three friction states: boundary-layer friction, semi-dry friction and liquid friction.

In Fig. 1 the boundary-layer friction is represented by section a–b of the curve, where the coefficient of friction has an almost constant value f_0. This state prevails when the oil film is very thin (molecular thickness) and the speed of the machine very low. As the speed increases, the coefficient of friction rapidly drops, section b–c. The oil film becomes thicker, but the peaks of the sliding surfaces will still be in contact with one another (semi-dry friction). The coefficient of friction reaches its minimum at point c, where the oil film is so thick that it just covers the surface irregularities. To the right of point c, section c–d, the coefficient of friction increases slowly as the value of $\eta\omega/p$ rises. A state of complete liquid friction prevails there. The surfaces are completely separate from one another, and the coefficient of friction depends on internal forces in the oil film. Since the heat developed in the bearing rises with increasing coefficient of friction, point c ought to correspond to the theoretically optimum conditions for the running of the bearing. There does not exist, however, any margin of safety at this point. The least reduction in the viscosity or speed will cause the peaks of the sliding surfaces to come into contact with one another. The total amount of heat developed at the point of contact will be very large, the oil is vaporised and new peaks will come into contact with one another. The process continues progressively towards point b, and the bearing temperature rises. To the right of point c, on the other hand, the conditions are stable. As an example assume that the estimated service point lies at point k. Displacement of this to the right results in an increased coefficient of friction. The heat developed increases, the viscosity of the oil drops and the service point is displaced to the left, closer to the estimated value. The service point is similarly returned towards the estimated value for a displacement to the left. The coefficient of friction then decreases, resulting in a reduced development of heat and increased oil viscosity.

The service point must therefore lie to the right of c and at such a distance from c that there is a sufficient margin of safety (the thickness of the oil film increases with increasing value of $\eta\omega/p$). The decisive factors in the choice of permissible values for the thickness of the oil film are, apart from the margin of safety, the machining accuracy and the load and heat deformations. The oil film must in fact be sufficiently thick so that it covers all irregularities in the sliding surfaces.

The dimensions of the bearing and the oil viscosity are therefore selected so that complete liquid friction prevails at the rated speed (the thickness of the oil film must not drop below the permissible value). When a machine is started and stopped, its speed is low. Consequently, the oil film is so thin that the sliding surfaces will be in metallic contact with one another—curve section b-c or, possibly, a-b according to Fig. 1. It is consequently the starting and stopping that cause bearing wear. To reduce the wear, the material for the sliding surfaces must be selected so that the coefficient of friction for metallic friction will be as small as possible.

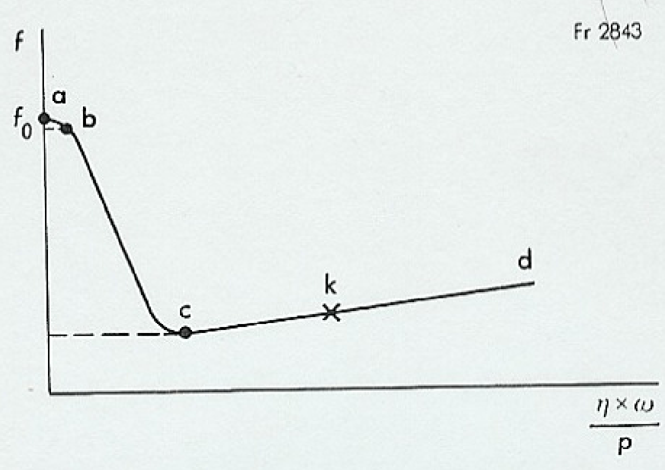

Fig. 1. Coefficient of friction, f, as a function of the dimensionless quantity $\eta\omega/p$, where η is the dynamic viscosity, ω the angular velocity and p the mean pressure.

THRUST BEARINGS

Design

Fig. 2 shows a cross-section of an ASEA thrust bearing combined with a guide bearing. The thrust bearing consists of a rotating disc (runner) 6 and a number of stationary pads 16 arranged on a large number of carefully manufactured helical springs 17, which are not precompressed. The bearing will thus be self-adjusting and the load is uniformly distributed over all the pads. Owing to the large supporting surface formed under the pads, these can be made relatively thin. The deformations will therefore be small and the pads easy to handle. This method of arranging the pads has been used by ASEA with excellent results since the beginning of the 1920's.

The sliding surfaces of the pads are coated with a thin layer of Babbitt's metal having a tin base, which has a low coefficient of friction for the starting and stopping. The bearing wear will be low and the machines can be started without it being necessary to jack up the rotor.

The runner is cast in steel in one piece without any separate disc and is shrunk on to the shaft. The fit is selected so that the runner can be easily mounted and removed by means of the special tool always supplied with the bearing. The load is transmitted from the shaft to the runner via a split thrust ring 5.

The bearing and oil reservoir shown in Fig. 2 are designed as a separate unit and can be arranged on the bearing bracket or on the turbine cover, depending on the design of the set. In machines with upper thrust bearing, where the latter may be subject to shaft currents, the bearing is insulated from the underlying surface.

The oil reservoir 26 has generous covers 29 to permit inspection of the bearings. When necessary, the bearing pads can be easily withdrawn through these covers for inspection.

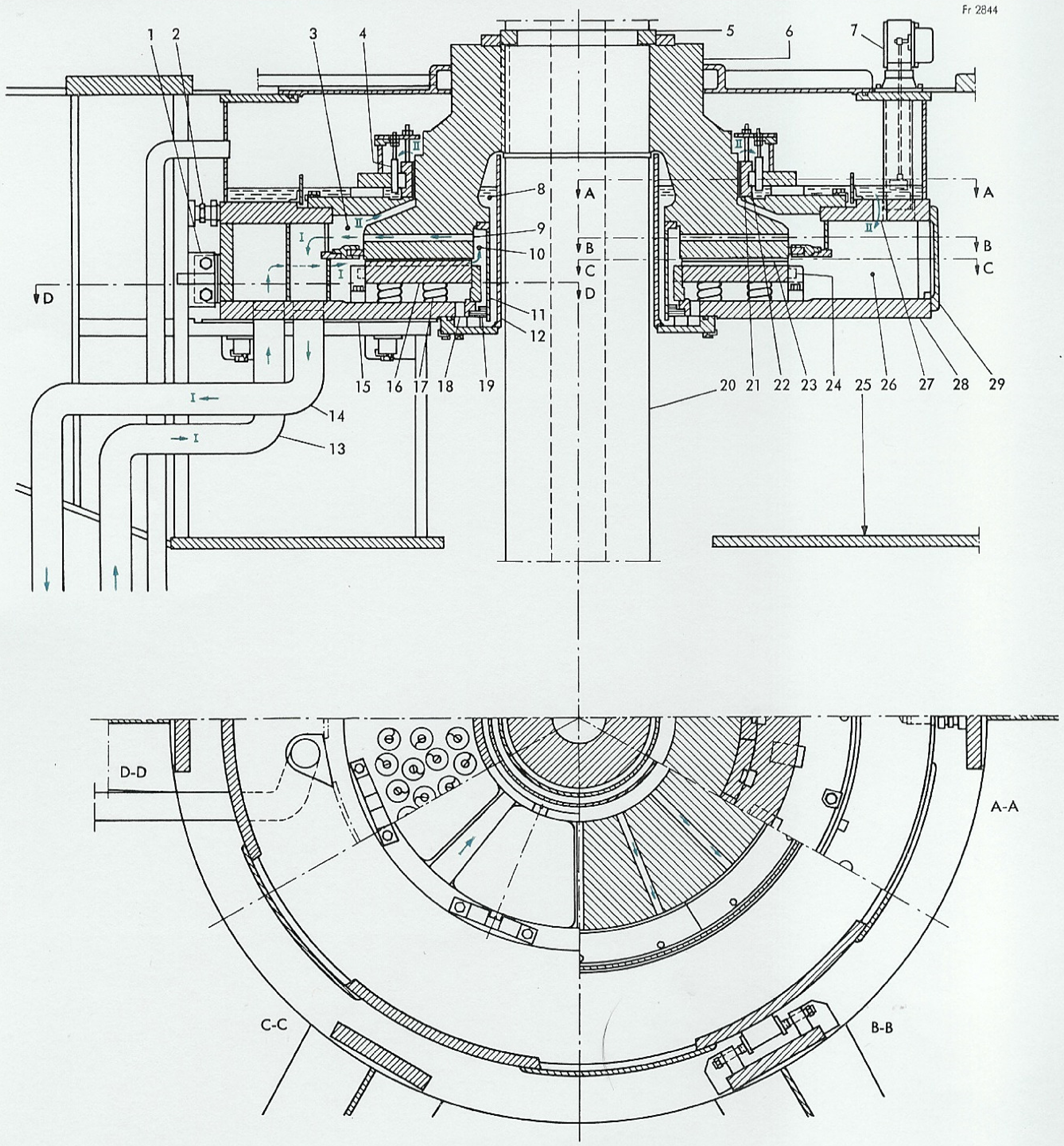

Fig. 2. Thrust bearing combined with guide bearing.

1. Tangential support
2. Setting screw
3. Oil-pressure chamber
4. Adjusting wedge
5. Thrust ring
6. Runner
7. Oil-level indicator
8. Inner oil space
9. Radial pumping duct
10. Oil intake chamber
11. Oil seal sleeve
12. Oil cup
13. Oil pipe from cooler
14. Oil pipe to cooler
15. Yoke with insulation
16. Thrust-bearing pad
17. Thrust spring
18. Pressure-equalising holes
19. Seal
20. Shaft
21. Guide-bearing pad
22. Pivot peg
23. Oil feed slot
24. Pad-retaining peg
25. Bracket
26. Oil reservoir
27. Guide-bearing oil-return holes
28. Thrust-bearing frame
29. Inspection cover

Oil circulation

The circulation of oil through the bearing and the external coolers is effected by a centrifugal pump arranged in the runner. The lubrication system does not have any separate oil pumps and is not therefore subject to any wear. The flow of oil from the pumping ducts 9 is distributed in two parallel branches, denoted I and II in Fig. 2. Branch I is arranged as follows: pressure chamber 3, cooler pipe 14, cooler (outside the bearing), cooler pipe 13, oil reservoir 26. Branch II is arranged as follows: pressure chamber 3, guide-bearing pad 21, hole 27, oil reservoir 26. The oil ducts in the guide bearing are designed so that the quantity of oil flowing through them meets the requirements of the guide bearing. This quantity of oil is usually very small in comparison with the total flow of oil. The cooled oil from branch I is mixed in the oil reservoir 26 with the guide-bearing oil from branch II and continues to the thrust bearing, where it flows radially into the slots 23 between the pads. The sliding surface of the bearing is thus lubricated and cooled. Oil flows from these ducts to the suction side of the pumping ducts 9 and through these to the pressure chamber 3.

Since the pump is of the centrifugal type, it does not have any suction capacity should the pump ducts be filled with air. It is extremely important, therefore, for the proper functioning of the bearing that no air is introduced into the oil circulation system. One point where air may penetrate direct into the pumping ducts is the seals around the shaft on the inside of the bearing. It is always difficult to obtain a perfect seal between rotating and stationary parts. A small quantity of fluid will always leak out from a space with higher pressure into a space with lower pressure. The seals against the shaft are arranged so that the oil itself acts as a seal.

As can be seen from Fig. 2, air must be prevented from being sucked into the chamber 10 from which oil is conveyed into the pumping ducts. The chamber 10, where an under-pressure prevails, is formed partly by the rotating sleeve 11 and the stationary seal 19. Yet another space 8 has been arranged around the chamber 10, where the oil has atmospheric pressure and is in communication with the oil reservoir 26 through the hole 18. The oil space 8 and the oil reservoir 26 thus stand in communication with one another and the oil levels will be at the same height. The oil in space 8 forms the seal to the shaft 20. A small quantity of oil leaks out from this chamber through the seal 19 to the chamber 10. This quantity of oil is replaced by oil from the reservoir 26 through gravity. A complete seal against air is thus obtained, which is a necessary condition for the proper functioning of the pump.

The self-circulation system described above for the lubricating oil has been successfully used by ASEA on all thrust bearings since the beginning of the 1940's.

Oil coolers

The external oil coolers are arranged according to the wishes of the customer and are coupled to the oil reservoir 26 via the pipes 13 and 14. They are designed so that one cooler can always be disconnected and cleaned during service. The water circuits can be cleaned with a suitable brush, after one end cover has been taken off without the oil cooler having to be removed from its foundation.

Hydrostatic lift

Machines which are subject to frequent starting or have a high bearing pressure are eventually liable to bearing wear. This can be avoided through the adoption of hydrostatic lift. Oil is then forced under high pressure into a recess on the middle of the sliding surface of the pad. The oil pressure is so high that the rotor is raised slightly and the sliding surfaces are lubricated. The pump set is switched on automatically for both starting and stopping operations. The oil inlet has a non-return valve, which prevents the middle of the bearing pad from being unloaded during normal service when the pump set is disconnected. If necessary, however, the machines can be started without hydrostatic lift. A pump set and thrust bearing pad designed for high-pressure lubrication are shown in Figs. 9 and 10.

Oil purification set

On request from the customer, the bearing can be provided with connections for an oil purification set. It is very important that this set as well as its valves are of such a design and are looked after in such a way that no air can be introduced through them into the oil circulation system of the bearing.

Design calculations

The bearing dimensions are determined from the following formulae and diagrams, which are solutions to Reynolds' equation for the hydrodynamic bearing theory.

Notation:

η viscosity of oils, Ns/m^2
v mean velocity, m/s
p mean pressure, N/m^2
l mean length of pad, m
b width of pad, m
F total load, N
h_{min} thickness of oil film at trailing edge, m
h_{max} thickness of oil film at entering edge, m
P frictional losses, W
$\varkappa$ $= b/l$
k $= (h_{max} - h_{min})/h_{min}$

The minimum oil film thickness at constant oil viscosity will be

$$h_{min} = K_F \cdot \sqrt{\frac{\eta v l}{p}}$$

and the frictional losses for the bearing face

$$P = K_P \cdot \sqrt{\frac{\eta v}{p l}} \cdot F v$$

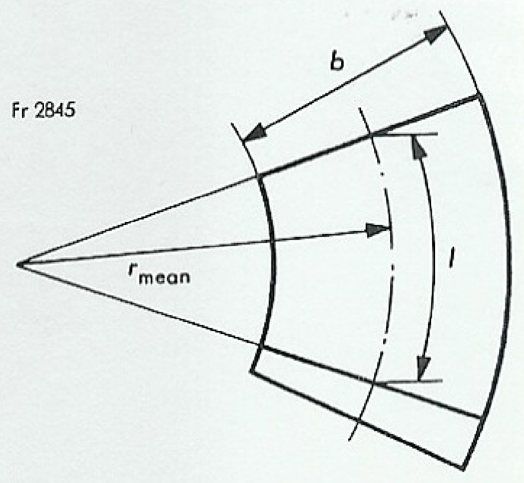

Fig. 3.

The dimensionless coefficients K_F and K_P are functions of the ratios $\varkappa = b/l$ and $k = (h_{max} - h_{min})/h_{min}$. These are plotted in Fig. 4 for k=1 and 2.

An optimum bearing must have the largest possible load-carrying capacity and the minimum bearing losses. To achieve this, the bearing must be designed for the correct values of $\varkappa$ and k.

In Fig. 5 K_F and K_P have been plotted as functions of k for $\varkappa$ =0.75, 1.0 and 1.5. From the diagram it is apparent that the oil film thickness (K_F) will have its maximum value for a k-value slightly larger than unity. The frictional losses K_P decrease with an increasing k-value.

As can be seen, the curves for K_F are rather steep to the left of the maximum value. If the bearing is designed for k=1, a small error in the alignment of the supporting point of the pad will result in a substantial reduction in the thickness of the oil film and a great increase in the losses. To the right of the maximum value the curves level out and any deviation from the k-value used in the calculations does not result in such large variations in the thickness of the oil film and losses. An increase of k from the value corresponding to the maximum value of K_F to 2 leads to a reduction in the thickness of the oil film of only 2–3 per cent. At the same time, the frictional losses are reduced by 5–7 per cent. In view of the above factor, the thrust-bearing springs for non-reversible machines are arranged so that the position of the resultant supporting force corresponds to k=2.

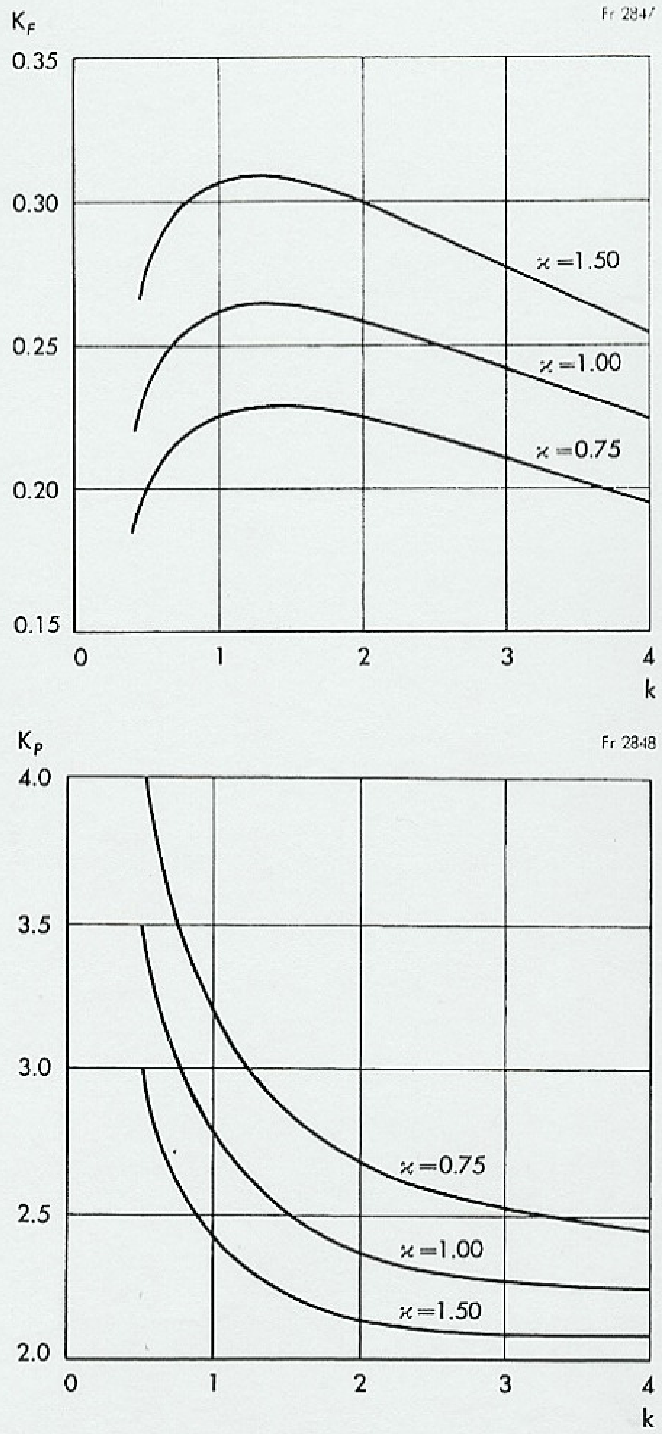

Fig. 5. The coefficients K_F and K_P as a function of $k=(h_{max}-h_{min})/h_{min}$

To investigate how changing of the b/l-value influences the thickness of the oil film and frictional losses, the length, l, of the pads is varied with unaltered pad width b. As the length is reduced so is the number of pads increased, and therefore the mean pressure will not be altered. The term $l = b/\varkappa$ is introduced in the formulae for h_{min} and P, giving

$$h_{min} = \frac{K_F}{\sqrt{\varkappa}} \cdot \sqrt{\frac{\eta v b}{p}}$$

$$P = K_p \cdot \sqrt{\varkappa} \cdot \sqrt{\frac{\eta v}{p b}} \cdot F v$$

For constant values of η, v, b, p, and F the thickness of the oil film and the frictional losses will depend on the factors $K_F/\sqrt{\varkappa}$ and $K_p \cdot \sqrt{\varkappa}$, respectively. These have been plotted in Fig. 6 as functions of $\varkappa$ for k=1 and 2. The largest thickness of the oil film and the minimum frictional losses are obtained for a value of $\varkappa$ which is slightly less than unity.

The coefficient of the temperature rise in the oil film has been plotted in Fig. 7 as a function of $\varkappa$. As can be seen, low values of $\varkappa$ result in a substantial temperature rise in the oil film. The temperature measured at the trailing edge may be high. For this reason ASEA's thrust bearings are usually designed for $\varkappa$-values lying in the vicinity of unity. In certain cases, when the pads will be exceptionally large and difficult to handle, $\varkappa$-values of up to 1.2 are applied.

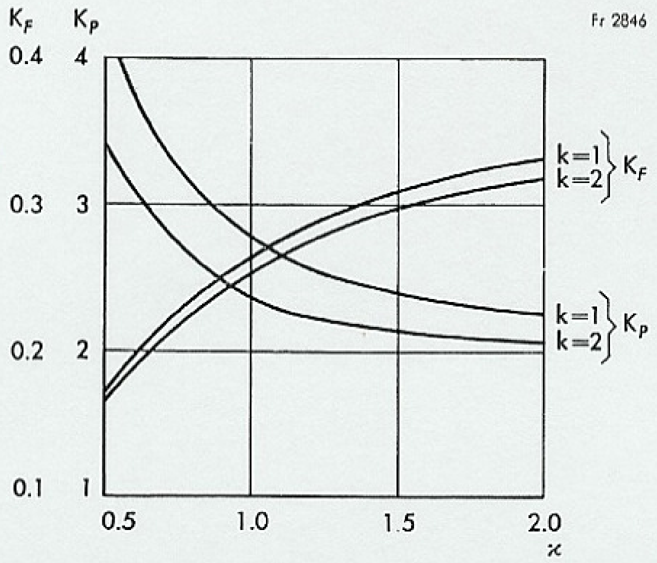

Fig. 4. The coefficients K_F and K_P for calculating the minimum thickness of the oil film and the frictional losses.

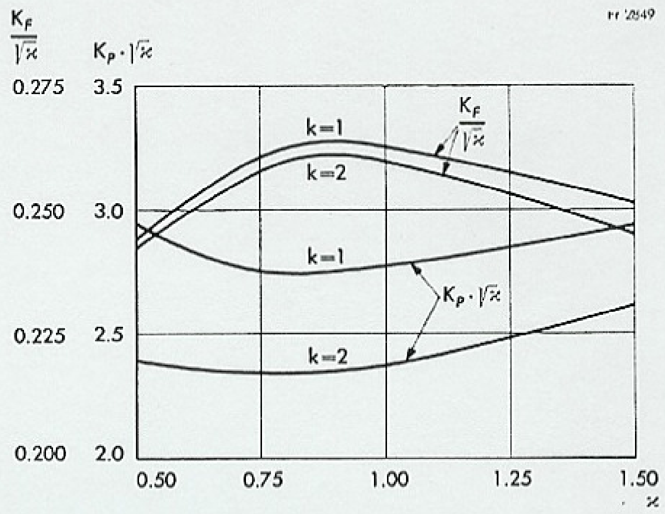

Fig. 6. $K_F/\sqrt{\varkappa}$ and $K_P\sqrt{\varkappa}$ as functions of $\varkappa = b/l$.

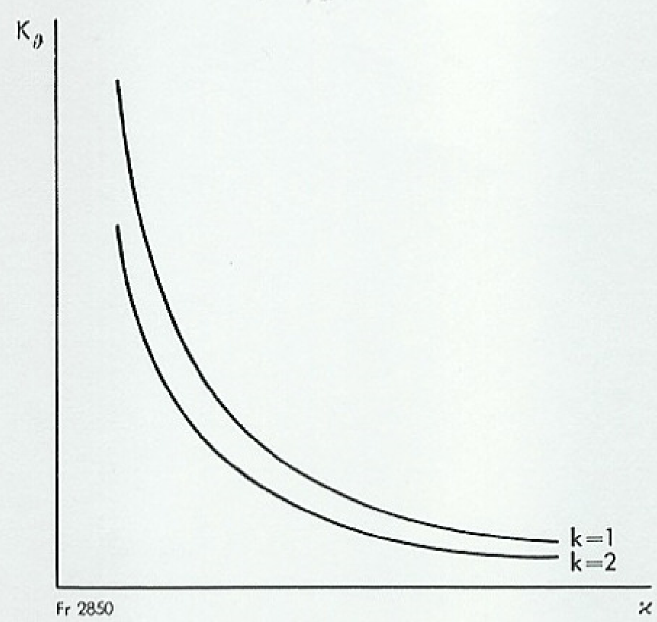

Fig. 7. Temperature rise in the oil film, $\Delta\vartheta = K_\vartheta p$.

Thickness of oil film and bearing temperature

As has already been mentioned, the reliability of the bearing depends on the thickness of the oil film formed during actual service. ASEA's thrust bearings are designed so that a nominal value of the thickness of the oil film is obtained at the normal speed for a certain oil temperature. This oil temperature is determined by the heat balance between the bearing losses and the dissipated heat. In this way, the bearing temperature will be fixed already at the design stage and can be said to constitute a "measure" of the thickness of the oil film.

The temperature is measured at the trailing edge of the bearing pads (hottest spot) as close to the sliding surfaces as possible. A higher bearing temperature than the calculated value thus implies a thinner oil film. Because of the uncertainties involved in the calculations, manufacture and temperature measurements, nominal thicknesses of the oil film are selected with such a margin that the bearing will function fully satisfactorily also at considerably higher temperatures than the calculated value. The nominal thicknesses of the oil film applied by ASEA vary between 30–60 μm, depending on the size of the bearing pad.

Bearing losses

The bearing losses in a thrust bearing are made up of frictional losses between the sliding surfaces, which are determined with the aid of the diagram in Fig. 4, and supplementary losses. The latter comprise the losses between rotating and stationary parts, sealing losses and pump losses. These losses usually amount to 20–30 per cent of the frictional losses. Since the supplementary losses constitute a rather large proportion of the total bearing losses, it is very important that these be included and calculated with sufficient accuracy for the determination of the capacity of the coolers. An insufficient cooler capacity will lead to a higher bearing temperature and a thinner oil film. The method of calculation applied by ASEA for the frictional and supplementary losses has been verified by means of careful calorimetric measurements in power stations.

The design of a separate guide bearing is shown in Fig. 8. This bearing is of the pad type with the pads 2 resting on eccentrically located bosses so that an individual oil film is created by each pad. The bosses rest against axially adjustable wedges 3, which permit an accurate setting of the bearing play. The circulation of oil is effected by means of a "viscosity" pump, which is formed between the rotating and the stationary parts of the bearing. If the bearing losses are so large that separate cooling is required, coolers 5 of the plug-in type are used. These coolers, which are built together with the bearing, can be easily replaced and the water circuits can be cleaned without the oil reservoir having to be emptied. Since the bearing is fully self-supporting with respect to the oil circulation and cooling, and no parts subject to wear are used, no supervisory devices are required except the thermometers incorporated in the pads.

A guide bearing combined with a thrust bearing is shown in Fig. 2. This guide bearing is of basically the same design as the separate guide bearing, but the oil circulation and cooling are associated with the thrust bearing, as has already been described in the chapter on thrust bearings.

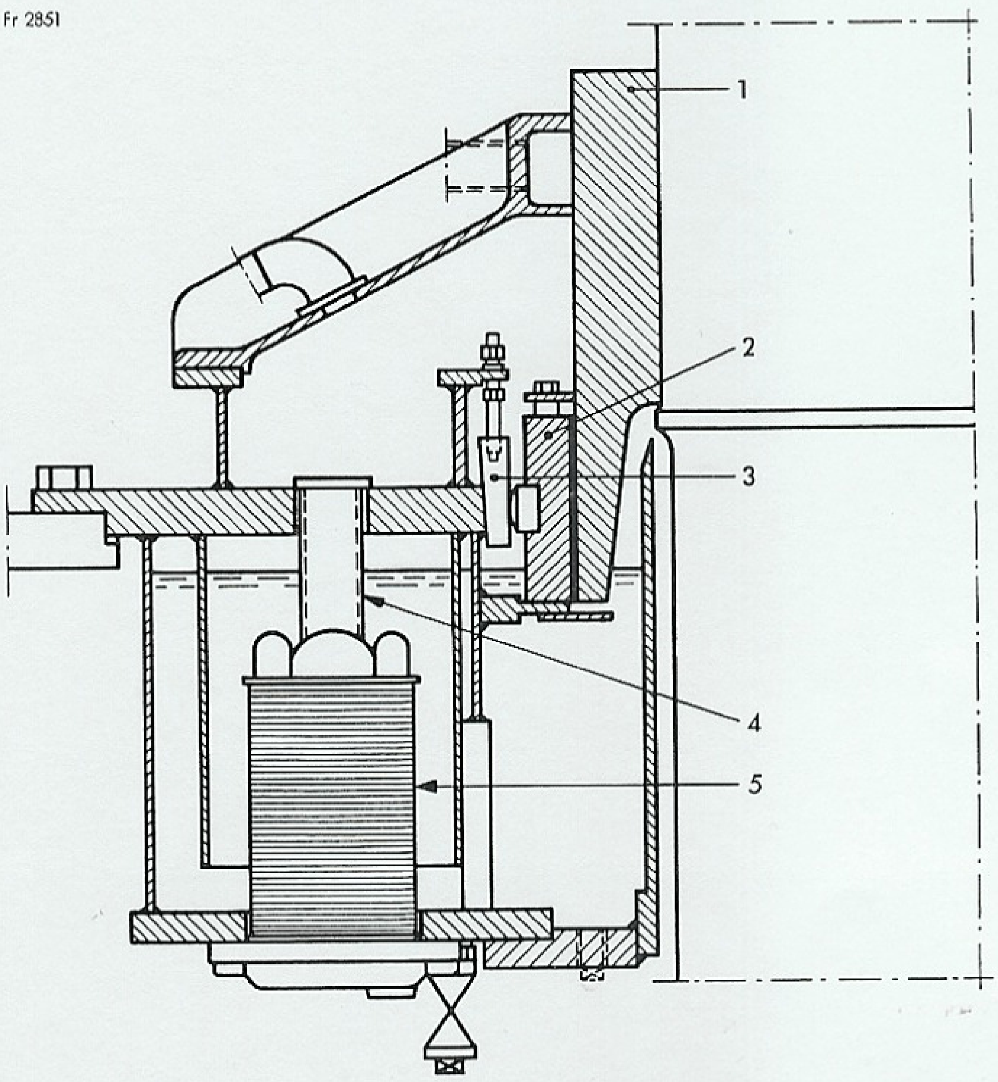

Fig. 8. Guide bearing.
1. Guide-bearing sleeve
2. Guide-bearing pad
3. Wedge
4. Oil pipe
5. Oil cooler

CONTROL EQUIPMENT

ASEA's thrust bearing are provided with the following control equipment to permit the supervision of the bearing during starting of the set and for continuous supervision during service: thermometers for measuring bearing temperature and oil temperature, pressure gauges for measuring oil pressure in the pressure chamber and level indicator for the oil in the oil reservoir.

As has already been mentioned, the bearing temperature is a "measure" of the thickness of the oil film to the extent that an increased bearing temperature implies a thinner oil film. The bearing temperature corresponding to the nominal thickness of the oil film for the size of pad used is calculated already at the design stage. This temperature is then used during the commissioning and service as a reference value for the assessment of the thickness of the oil film and the reliability of the bearing.

The nominal value of the thickness of the oil film at this reference temperature is selected with such a large margin of safety that the bearing will function fully satisfactorily also at a considerably higher bearing temperature. This continuous following up of the bearing and oil temperatures during service is intended in the first place to provide a check that no gradual change is taking place in the functioning of the bearing. When this is being assessed, it is necessary to take into account that the temperature varies with the load and the temperature of the cooling water. The thermometers are wired up for automatic indication if the set temperatures are exceeded so as to facilitate this following up of the temperatures. The thermometer for measuring the bearing temperature is also connected for automatic tripping of the set. The tripping temperature is set to a few degrees above the signalling temperature.

It should be noted in this connection that neither the signalling temperature nor the tripping temperature are to be considered as limiting values, where the reliability of the bearing is jeopardised. The temperature at which the oil film becomes so thin that it may be broken down lies far above these values.

The oil circulation in the bearing is monitored with the pressure gauge connected to the pressure chamber. The normal quantity of oil circulating through the bearing and coolers corresponds to a certain oil pressure in the pressure chamber. A change in the oil circulation thus results in an altered oil pressure (e.g., a leakage by the seals against the runner leads to a reduced oil pressure in the pressure chamber). During the commissioning the oil pressure in the pressure chamber is checked to see that it corresponds to the calculated pressure value. The continuous monitoring in the form of pressure measurements during service is supplemented by a pressure gauge, which gives a signal if the oil pressure drops below the set value.

The level indicator, which shows the oil level in the oil reservoir, is connected so as to give a signal for both insufficient and excessive oil level. A signal is thus obtained for any leakage in the oil system of the bearing.

It should be readily apparent from the above that the control equipment for the bearing serves two purposes: signalling and tripping. Signalling provides information about a slow change in the functioning of the bearing, while tripping automatically stops the set. In both cases, the reason for the fault indication must be investigated and the defect remedied.

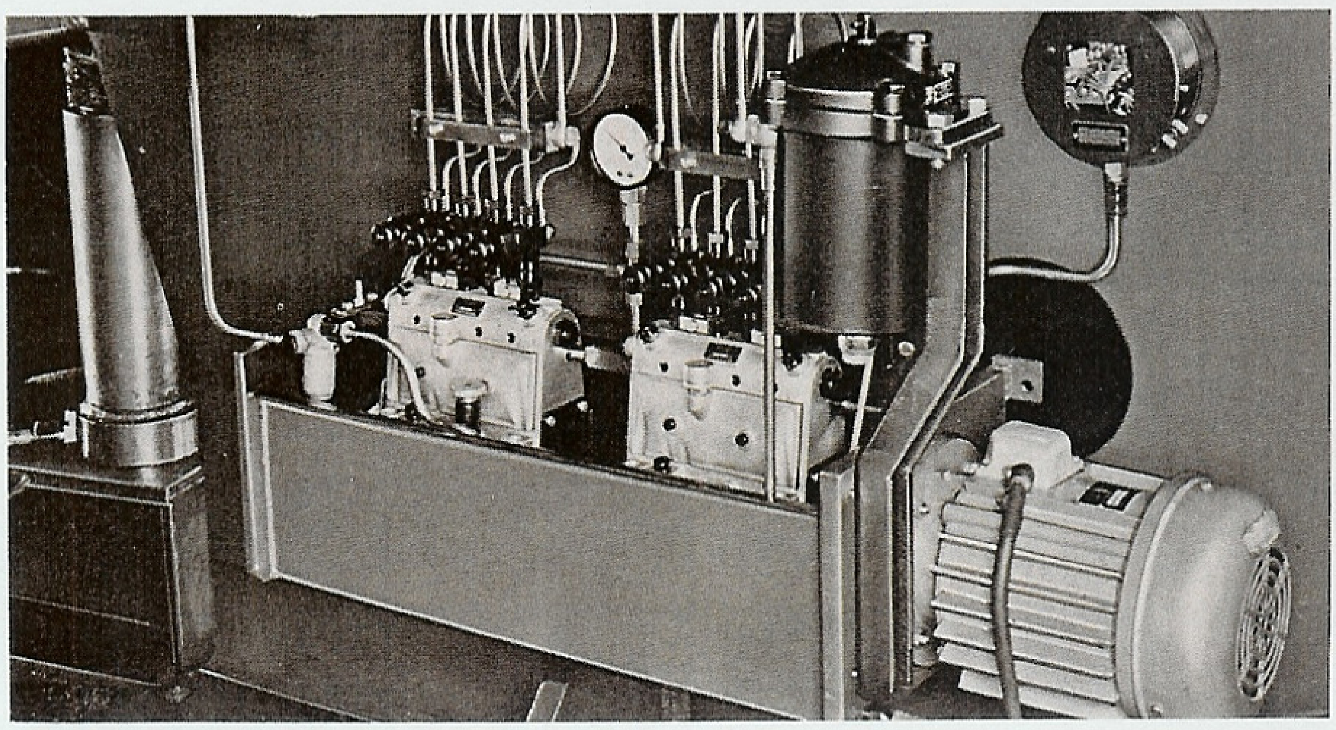

Fig. 9. Oil pump set with reciprocating pumps for hydrostatic lift. (57507)

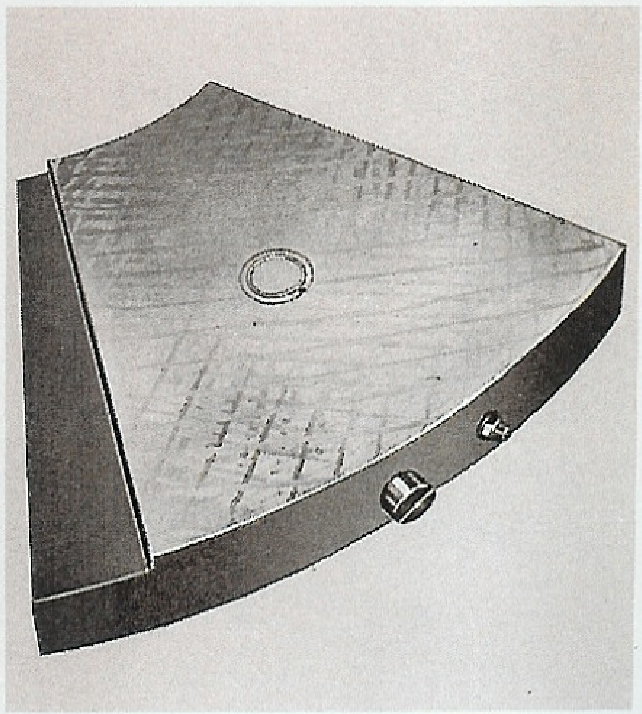

Fig. 10. Thrust-bearing pad with recess for pressurised oil. (57385)

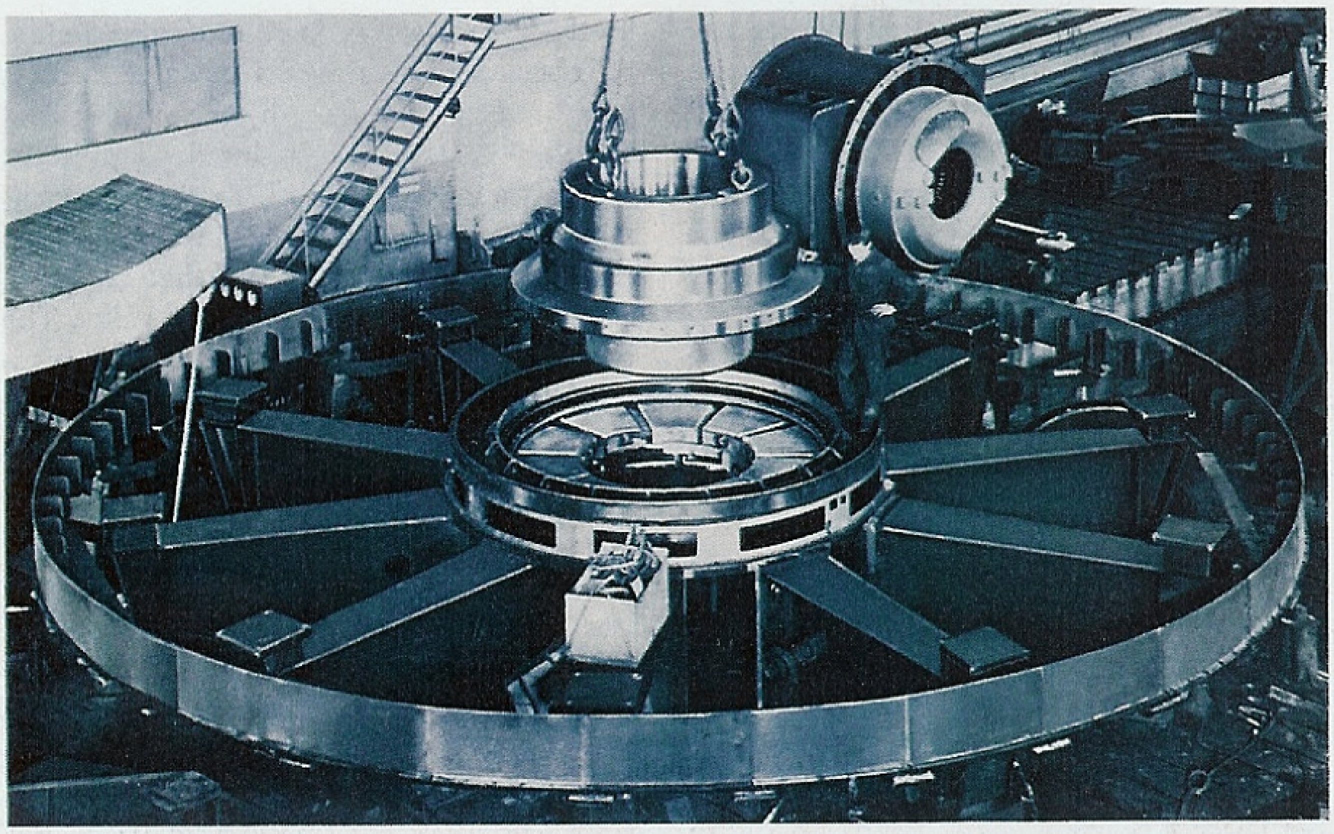

Fig. 11.

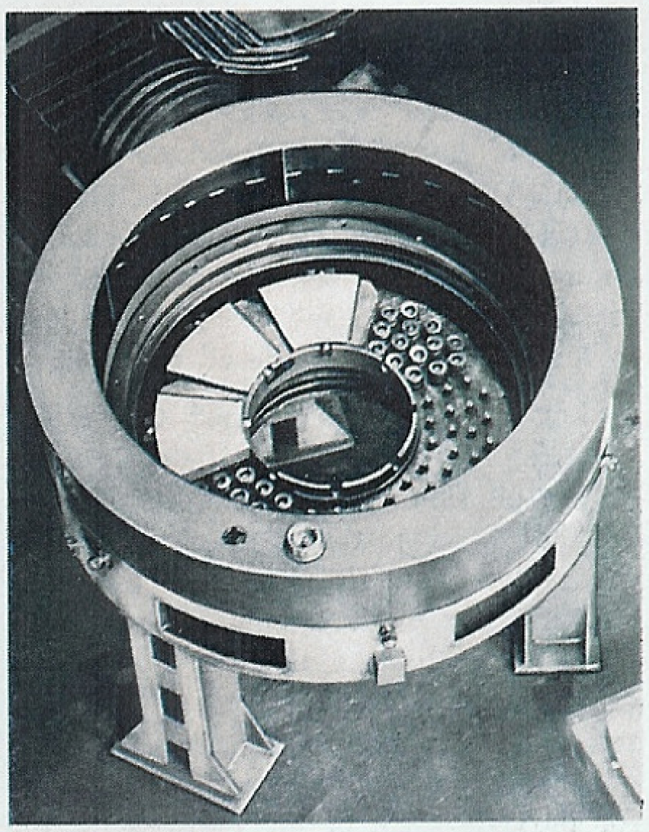

Fig. 12.

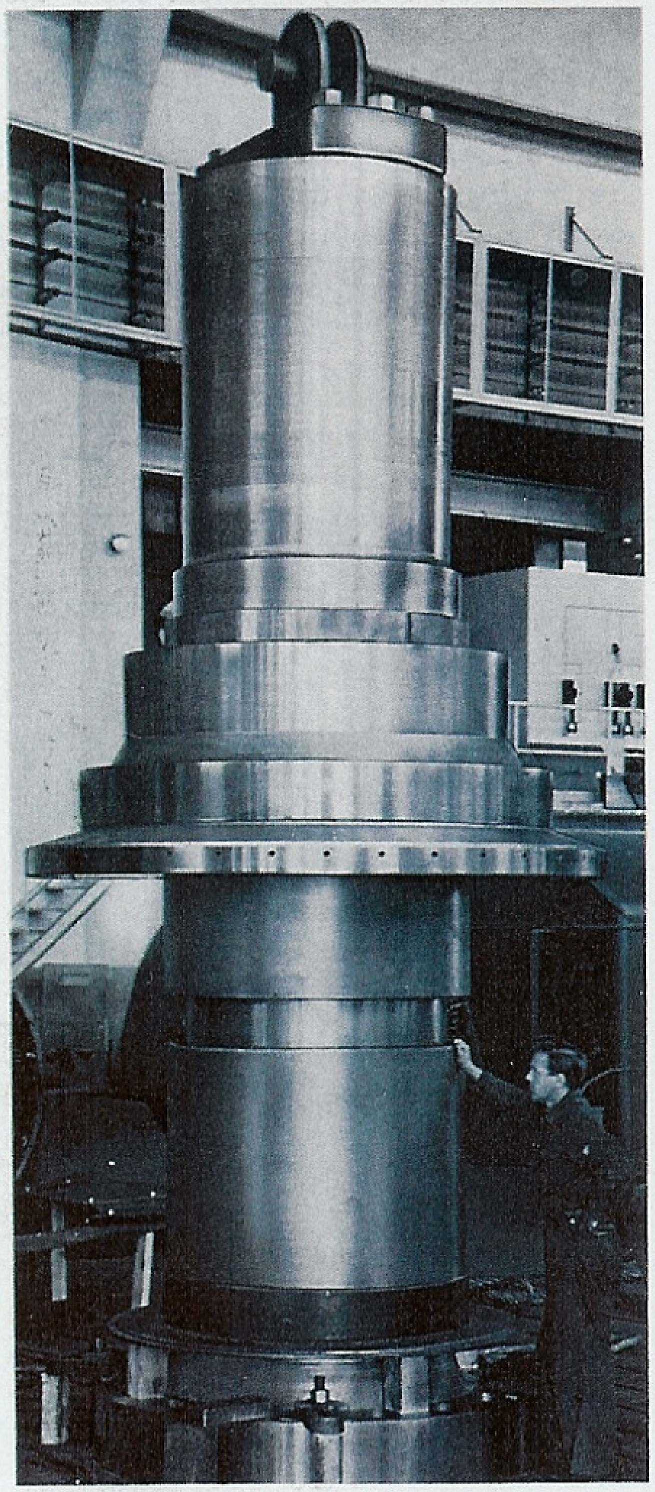

Fig. 13.

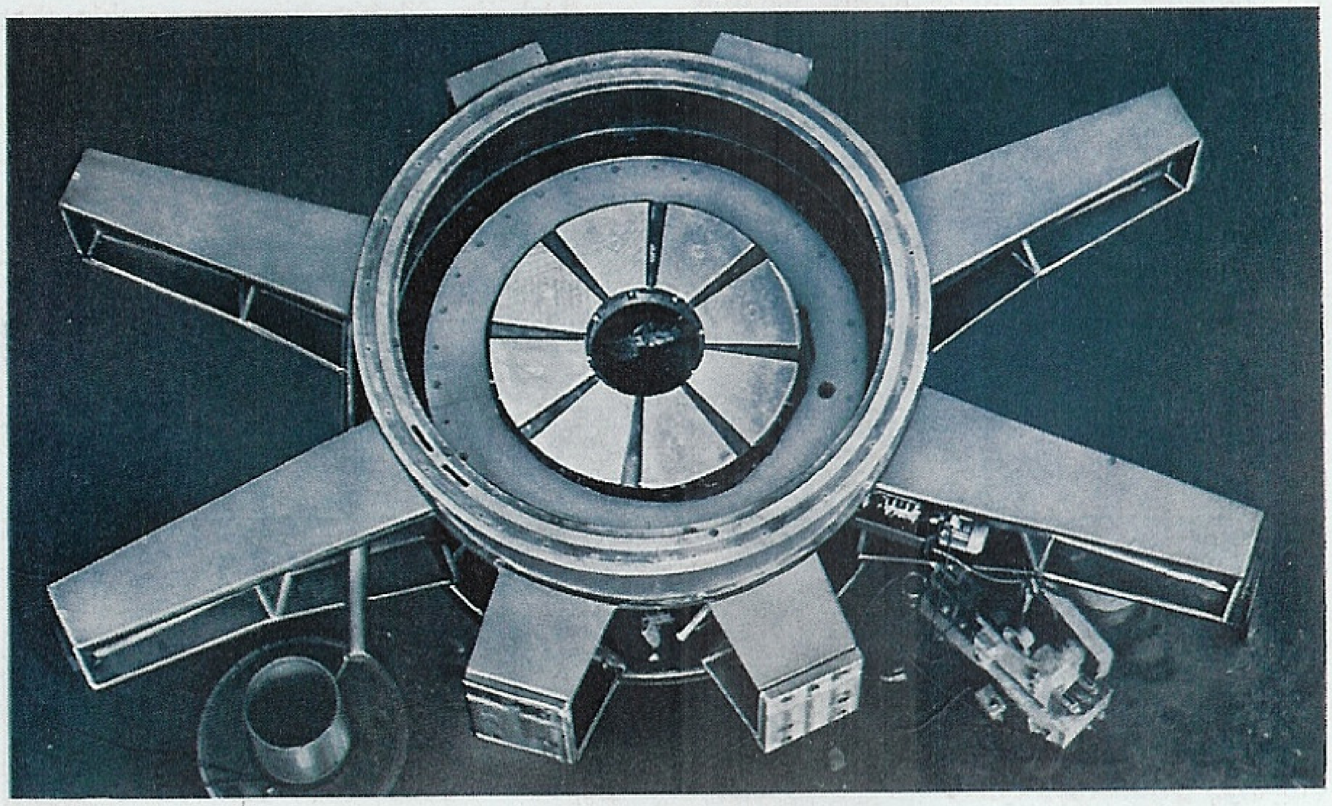

Fig. 14.

Fig. 11. Thrust bearing with runner raised. (51494)

Fig. 12. Thrust-bearing frame showing a few bearing pads and springs in position. (61741)

Fig. 13. Generator shaft with thrust-bearing runner ready for insertion in bearing bracket. (51659)

Fig. 14. Thrust-bearing assembly with pads mounted in bearing bracket. (57510)

Fig. 15. Thrust-bearing spring. (67260)

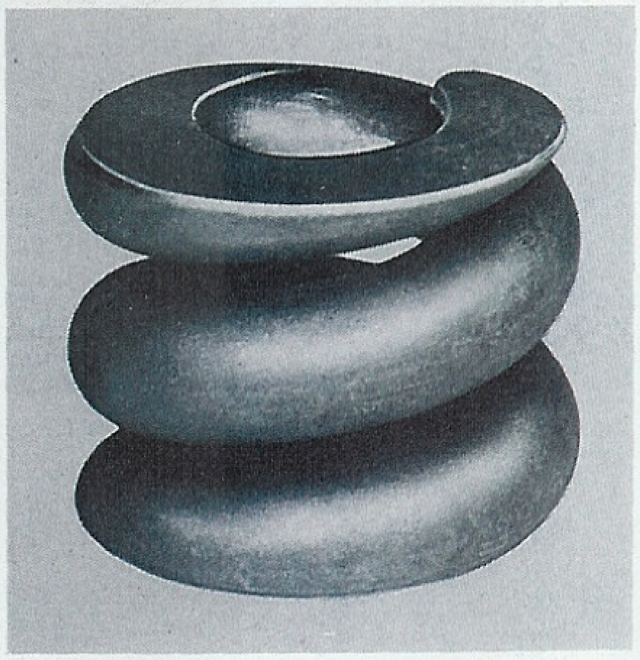

Fig. 15.

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